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Development of a highly compact steam generator

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Abstract

Conventional steam generators are the largest components in steam power plant, and add considerably to the physical size of gas turbine based combined cycle and combined heat and power plant. Availability of compact steam generators capable of operating at high pressures would make heat recovery from engine and turbine exhausts attractive in a wider range of applications. The advantages of small size are particularly marked in mobile systems.

This paper describes the design and preliminary testing of a steam boiler suitable for use in a steam engine powered car. The boiler has a nominal rating (based on water entering at 100 °C when producing steam at 40 bar, 400 °C) of 135 kW. The volume of the boiler (excluding the burner) is approximately 0.06 m^3, giving a volumetric power density of over 2 MW/m^3. Mechanical design of the boiler is such that the operating temperature and pressure may be increased to improve overall system performance.

The influence of various parameters on the performance and stable operation of the boiler is examined. It is shown that the concepts discussed could be applied to heat recovery steam generators, thus permitting economical heat recovery from internal combustion engines and micro gas turbines.

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Nomenclature

n	number of increments
U	overall heat transfer coefficient, W/m^2 K
q	heat flux, W/m^2
Q	rate of heat transfer, W
d	tube diameter, m
T	temperature, °C
z	distance along tube, m
Δz	incremental length, m
p	pressure, bar, N/m^2
x	quality (dryness fraction)
G	mass flux
α	heat transfer coefficient, W/m^2 K

Subscripts

i	counter
g	gas side
s	steam/water side

1. Introduction

In the early years of the last century steam powered automobiles were competitive, both technically and commercially, with vehicles using internal combustion engines. Both types of vehicle had starting problems: the steam car needed a relatively lengthy period to raise steam; internal combustion engines were started using a hand crank. By the mid 1920s most IC engine powered cars were available with electric self starters and the popularity of the external combustion engine declined. Continuous development and refinement of the internal combustion engine, supported by enormous budgets, has resulted in enhancement of efficiency, power output and reliability. Meanwhile, efforts to develop small steam power plant have been sporadic. Advances in technology, most notably in electronic control and materials, many of which have been adopted by the automobile industry, have been largely untried in relation to small steam plant. Several projects which examined the potential of steam-powered automobiles in the 1960s and 1970s (e.g. [1–3]) preceded the widespread availability of micro-electronics and were thus unable to benefit from the sophisticated control which microprocessors now permit. An investigation into the likely benefits and performance of a modern steam car by the present authors and others commenced in 1998 [4].

The requirements of an automobile power plant impose unique constraints on the design of a boiler; however, a boiler designed for this purpose would have other potential applications, for example in compact micro-chip or combined cycle plant.

This paper describes the design and preliminary testing of the boiler.

2. Specification and design

After analysis of the power cycle and proposed engine characteristics, the performance specification listed in Table 1 was proposed for initial testing.

The following characteristics are desirable in a steam-car boiler:

- *Highly compact*: from the point of view of the user, space taken up by the power plant in a vehicle is of no value. Hence the boiler must be as compact as possible. The allowable dimensions for the prototype boiler described in this paper were chosen so that the boiler and burner would fit above the gearbox and behind the rear passenger seats of the concept car, based upon a Volkswagen Beetle [4]. Low mass is desirable, from the point of view of vehicle performance, for safety reasons and as a consequence of the requirement for low thermal capacity.
- *Low thermal capacity*: Rapid start up and response to changing loads demand a boiler having low thermal capacity.
- *Low fluid inventory*: The start up time increases with increasing fluid inventory. More importantly, reducing the fluid inventory also reduces the consequences of unintended release of high pressure steam–water mixture in the event of boiler failure or accident.
- *Safe venting*: The design of the boiler must be such that steam may be safely vented if a tube fails or pressure relief device opens.
- *Ease of maintenance*: conventional steam boilers are subject to specialist maintenance procedures and inspection. If a boiler is to be incorporated in a mass produced product such as a car, it must be designed so that it can be maintained by less skilled personnel.

Taking into account the above considerations, a boiler was designed and built using parallel coils of stainless steel tube connected to inlet and outlet manifolds by compression fittings. The manifolds themselves were machined from solid stainless steel. The tube internal diameter was 6 mm with a 1 mm wall thickness. Use of compression fittings, rather than welded joints at the manifolds permitted a construction which was free from welds, and thus required no Non-destructive testing (NDT) after assembly, other than hydrostatic pressure test, and could be maintained without recourse to special equipment. Such a design is only acceptable for small bore tubes; leakage, or even complete failure of a compression fitting would lead to a relatively small escape of steam into the combustion chamber and through the exhaust. A bursting disc was incorporated in the water inlet line to provide overpressure protection. Discharge from the bursting disc was

Table 1
Boiler thermal specification

Parameter	Value
Water inlet temperature	100 °C
Steam outlet temperature	500 °C
Boiler pressure	40–70 bar
Saturation temperature	264 (@50 bar) °C
Steam flow rate	153 kg/h
Max. rate of heat transfer	135 kW
Burner nominal output	166 kW

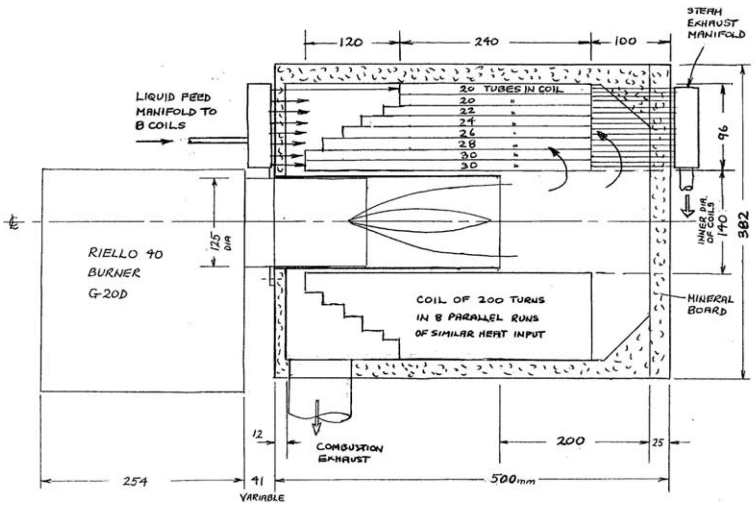

Fig. 1. General arrangement of the boiler.

vented into the exhaust from the combustion chamber. This arrangement proved satisfactory, rupture of the bursting disc during operation was observable firstly through obvious changes in operating parameters and then through a trickle of water from the exhaust pipe.

The general arrangement of the boiler is shown in Fig. 1. and key features are illustrated in the photographs of Fig. 2.

3. Heat transfer

A program was written to evaluate the heat transfer and pressure drop through a single coil of the boiler. Due to the complex nature of the flow on the gas side of the boiler and the irregular tube spacing after manufacture, it was not possible to accurately determine the gas side heat transfer heat transfer coefficient. A simple analysis, treating a single tube as a counterflow heat exchanger and using representative heat transfer coefficients in the subcooled water, boiling and superheated steam sections of the boiler showed that the governing heat transfer resistance would be expected to be that on the gas side throughout the boiler. The variation in steam side heat transfer coefficient along the tube and the calculated temperature profiles are illustrated in Figs. 3 and 4 respectively.

These calculations suggested that 8 parallel tubes of length of 23 m would provide adequate heat transfer.

4. Stability and flow distribution

The simplified heat transfer calculations which were used to determine the heat transfer area assumed that the flow was evenly distributed between the tubes and that the heat transfer was

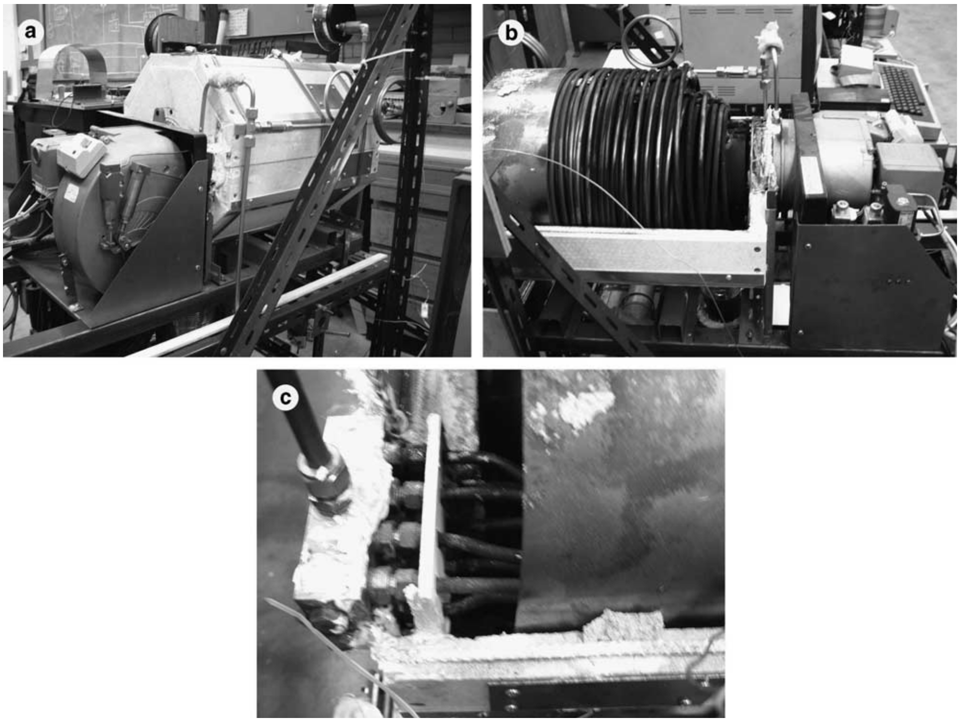

Fig. 2. Photographs of the boiler (a) (top left) boiler and burner, (b) (top right) boiler with upper casing removed, (c) (centre) outlet manifold showing compression fittings and heat shield.

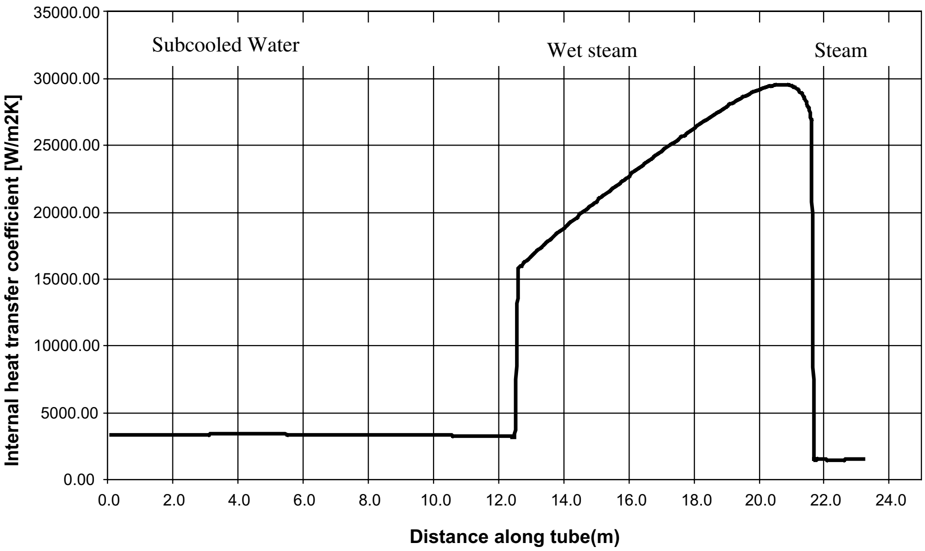

Fig. 3. Calculated water/steam side heat transfer coefficient.

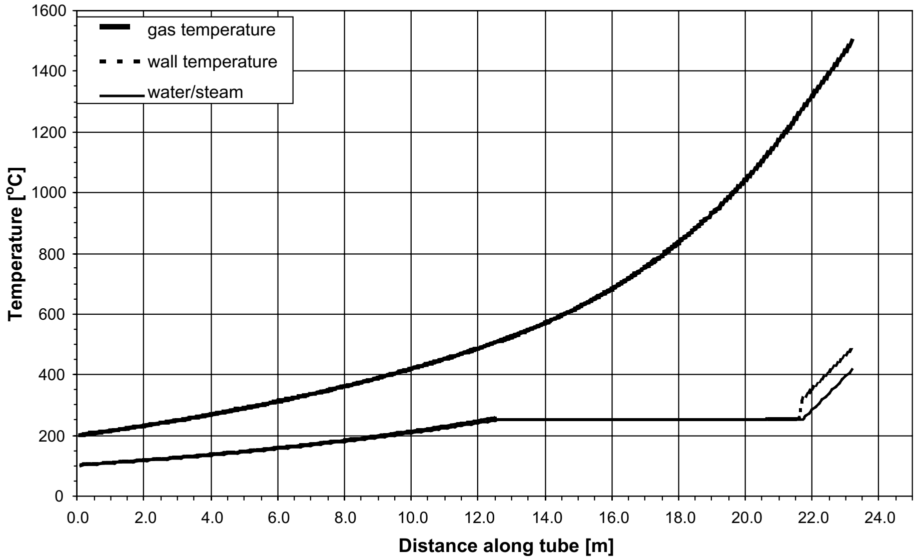

Fig. 4. Temperature distribution along length of tube ($\alpha_{gas} = 80$ W/m^2 K).

identical for each tube. Further calculations were carried out to assess potential problems of flow instability and mal-distribution.

It is well known that flow instabilities may occur when fluids boil in parallel tubes. The relationship between pressure drop and mass flow is extremely sensitive to tube length and gas-to-tube wall heat transfer coefficient, both of which vary from tube to tube in a boiler of the type described in this paper. Additionally, the Ledinegg instability [5] can lead to flow mal-distribution or instability in parallel, identical tubes.

An analysis was carried out to determine the extent to which various factors influence flow mal-distribution. For the purpose of this analysis, it was assumed that the gas temperature profile was independent of the performance of any individual tube. This would be true in a boiler with many tubes, in a boiler with only 8 parallel tubes large variations in the performance of one tube would have a measurable effect total heat transferred and hence on the gas temperature profile. The assumption that the gas temperature profile follows that of Fig. 4 will yield conservative results, in that the effect of changes in tube length and external heat transfer coefficient would be attenuated if the temperature profile was to be revised to take into account the change in heat transfer to the tube.

5. Outline of calculation procedure

A computer program was written in Visual Basic, embedded in Excel to calculate the heat transfer and to the water/steam in an individual tube and the pressure drop through the tube. The following parameters were set:

Tube diameter, wall thickness, thermal conductivity and length.
Water inlet condition.
Steam flow rate.
Gas inlet and outlet temperatures and temperature profile.
Gas side heat transfer coefficient.

The procedure for calculation may then be summarised; using the nomenclature of Fig. 5:

For $i = 1$ to n
Calculate local U
Calculate local q: $q_i = U_i(T_{g,(i-1)} - T_{s,(i-1)})$
Calculate ΔQ_i: $\Delta Q_i = q_i(\pi d\Delta z)$
Calculate $T_{g,i}$, $T_{s,i}$, x from heat balance
Calculate $\left(\frac{dp}{dz}\right)_s = f(G, x, \Delta Q_i)$

$$p_{s,i} = p_{s,(i-1)} + \left(\frac{dp}{dz}\right)\Delta z = p_{s,(i-1)} + \Delta p_i$$

$$Q = \sum_{i=1}^{i=n} \Delta Q_i$$

$$\Delta p = \sum_{i=1}^{i=n} \Delta p_i$$

This procedure was repeated for a range of steam mass flow rates wit various tube lengths and gas side heat transfer coefficients.

The calculation method used for determining the local tube side heat transfer coefficient and pressure gradient was dependent upon the condition of the water/steam. For liquid only flow ($x < 0$) and vapour only conditions ($x > 1$) the Dittus–Boelter equation was used to determine the heat transfer coefficient and the pressure gradient was calculated using a friction factor determined from a Moody diagram, approximations to the relevant e sections of which were included in the program. During evaporation ($0 < x < 1$) the Gungor and Winterton correlation [6] was

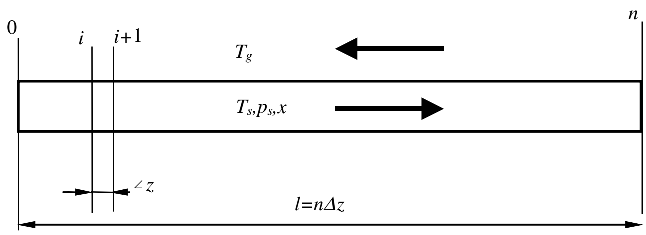

Fig. 5. Nomenclature used in calculation of heat transfer and pressure drop.

used to estimate the heat transfer coefficient and the pressure drop evaluation was based upon Lockhart and Martinelli as described in [7]. The slip factor was calculated using Chisholm's correlation [8].

6. Results of the calculations

The Ledinegg instability occurs when the pressure drop—mass flow curve exhibits a maximum in the two-phase region and a minimum value when the flow is such that saturated liquid exits the tube. This characteristic is shown for the compact boiler tube in Fig. 6. If the characteristic of the pump supplying a single tube is such that it crosses the tube pressure drop line in three points there are three possible operating conditions for the system. The compact steam car boiler was supplied by a positive displacement pump, therefore the total mass flow for all tubes was fixed. For a system comprising multiple channels with a fixed total mass flow, the unstable operating region would be as marked on Fig. 6. This indicates that this particular instability is unlikely to be problematical in a boiler designed to operate with a significant degree of superheating at exit. However, if the flow rate was to be increased the boiler could operate under an unstable condition such that the flow rate and outlet conditions varied significantly from tube to tube.

Geometric and constructional considerations make it impossible to construct a coils boiler such that all tubes have identical lengths. Predicted pressure drop characteristics for tubes of lengths ranging from 23–28 m are shown in Fig. 7. As expected, at high flow rate resulting in single phase liquid flow throughout, the pressure drop predicted is proportional to tube length. When operating in the two-phase region or with superheated steam at exit, the predicted pressure drop increases more steeply that the length. For parallel channels with identical pressure drop and

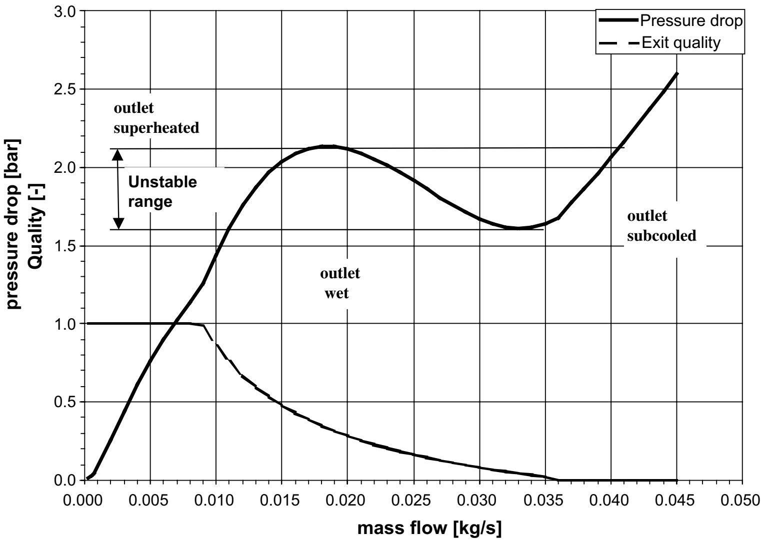

Fig. 6. Calculated pressure drop in tube ($\alpha_{gas} = 80\ W/m^2\ K$).

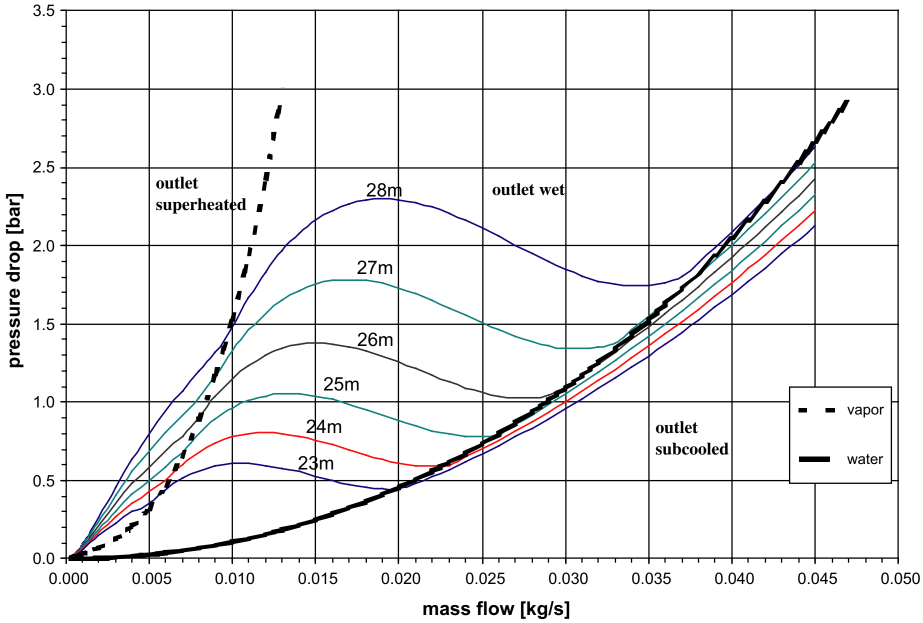

Fig. 7. Influence of tube length on pressure drop.

operating at low average flow rate with superheated steam at exit, the mass flow would be expected to decrease significantly and the superheat increase with increasing length.

As stated above, the gas side heat transfer coefficient cannot be accurately determined, and is likely to vary from tube to tube and along the length of individual tubes. To assess the magnitude of the influence of change in this coefficient pressure drop has been plotted against mass flow for a range of gas side heat transfer coefficients. It can be seen that, unsurprisingly, the magnitude of the gas side heat transfer coefficient has a major influence on the condition of the steam leaving the tube and on the pressure drop through the tube at any mass flow rate. For a fixed pressure drop, increasing heat transfer coefficient leads to a reduction in flow and increase in superheat or exit quality. It is reiterated that the calculations were performed on the basis that the gas temperature profile was unaffected by changes in gas side heat transfer coefficient. If all tubes in the compact boiler have the same gas side heat transfer coefficient the gas temperature profile will change and, given the relatively close design approach temperature at the water inlet—gas exhaust end of the boiler, the performance of each tube will be similar to that for a gas side coefficient of 80 W/m^2 K. If, however, the gas side heat transfer coefficient varies significantly from tube to tube, Fig. 8 shows that the flow rate and superheat at exit from each tube will differ greatly.

7. Use of flow restrictors

The results shown in Figs. 6–8 indicate that, at the design conditions, the pressure drop across the parallel tubes would be approximately 0.75 bar. In order to reduce the dependency of flow rate in tubes on pressure drop it is common practice to introduce flow restrictors at the inlet of each

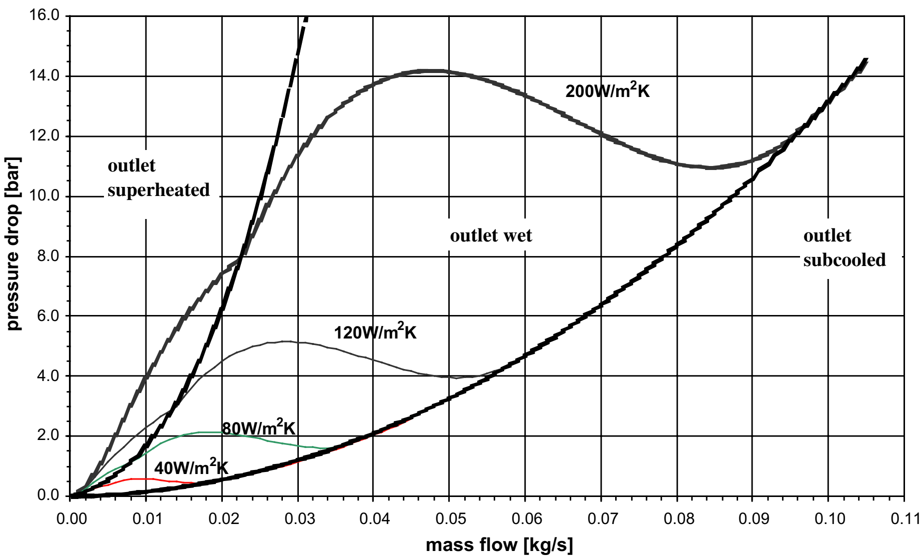

Fig. 8. Influence of gas side heat transfer coefficient on pressure drop.

individual tube. At the design conditions the pressure drop across the restrictor is higher than that across the tube, therefore perturbations of the conditions within the tube resulting in changes in pressure drop to not result in a large percentage change in the overall pressure drop across the tube and restrictor.

Initial commissioning tests on the compact boiler confirmed that the performance of individual tubes differed to an unacceptable degree. Addition of flow restrictors mounted in the inlet manifold of the boiler solved this problem. The restrictors consisted of 1 mm thick PTFE orifice plates with 1 mm diameter holes mounted in the manifold and located by the stud coupling holding the tube. Acceptable variation of temperature at the outlet of the tubes was noted in subsequent commissioning tests.

8. Test results

A very limited set of test results has been obtained. Commissioning tests were carried out to ensure that the boiler and burner functioned correctly and confirmed the rapid response of the boiler to changes in firing rate. Minor modifications were made during this phase of the project. The boiler was tested in conjunction with the steam engine. Problems with the engine prevented long term or high pressure operation of the boiler. A typical set of results is shown in Table 2. The burner was operating at approximately 80% of its maximum capacity and it can be seen that the boiler performed as expected, producing slightly superheated steam. A more comprehensive test programme is required to fully validate the calculations described in this paper, but it has been demonstrated that the boiler is suitable for producing steam for use in a steam powered car.

Table 2
Results of commissioning tests on compact boiler

Parameter	Value
Water inlet temperature	30 °C
Steam outlet temperature	236 °C
Boiler pressure	28 bar
Saturation temperature	230 °C
Steam flow rate	140 kg/h
Heat transfer to steam	105 kW
Exhaust temperature	186 °C

9. Conclusions

A highly compact water tube boiler has been designed and constructed for use in a steam powered road vehicle. In a highly compact boiler with multiple parallel tubes, variation in gas side heat transfer coefficient is likely to cause poor flow distribution between the tubes. However calculations and tests have shown that potential problems with flow distribution and stability may be overcome by incorporating flow restrictors in the inlet manifold.

Acknowledgement

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References

[1] R.M. Palmer, An exercise in steam car design, Proc. IMechE 184 (1969–1970) 195–218.
[2] S. Luchter, R.A. Renner, An assessment of Rankine engines for automobiles, Report ERDA-77-54, Division of Transport Energy Conservation, USA, 1977.
[3] J.W. Carter, The Carter System, SAE (USA), Report 750071, 1975.
[4] K. Cornwell, G.C. Hartnup, P.A. Kew, S.D. Houston, Heat transfer problems in modern steam car development, in: Proceedings of the Sixth UK National Conference on Heat Transfer, IMechE Conference Transactions 1999-7, 391397, 1999.
[5] P.B. Whalley, in: Boiling, Condensation and Gas–Liquid Flow, Oxford Engineering Science Series, vol. 21, Oxford Science Publications, Oxford, 1987, p. 116.
[6] K.E. Gungor, R.H.S. Winterton, A general correlation for flow boiling in tubes and annuli, Int. J. Heat Mass Transfer 29 (3) (1986) 351–358.
[7] K. Stephan, Heat Transfer in Condensation and Boiling, Springer-Verlag, Berlin, 1992.
[8] D. Chisholm, Void fraction inn two-phase flow, J. Mech. Eng. Sci. 15 (1973) 235–236.